14 Exhaust Heat Recovery

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14.1 Basics of Waste Heat Recovery

14.1.1 Preliminary Remarks

Although public awareness of the finiteness of fossil fuel reserves has receded into the background somewhat after being raised in the 1970s, the impact of pollutant and CO_2 input into the earth's atmosphere is again making the need for a longer range environmentally compatible energy policy with concrete goals evident.

In the future, both challenges – conserving resources and protecting the environment – will increasingly require an approach that endeavors to take full advantage of the ample potentials to save energy and additionally intensify the utilization of renewable, i.e. inexhaustible, energy sources. Both goals will have to be pursued simultaneously, i.e. in parallel, rather than sequentially.

This will necessitate research on the types of waste heat that accumulate during diesel engine combustion as well as expedient recovery methods for the purpose of conserving primary energy and protecting the environment.

14.1.2 Diesel Engine Waste Heat

The following types of waste heat can be distinguished on the basis of their origin:

- waste heat from exhaust gas generated by gas exchange,
- waste heat produced as cooling energy to protect metallic walls, e.g. cylinder cooling, piston cooling and, where applicable, cooling of turbocharger turbine housings and oil cooling of bearings and interior walls,
- waste heat from intercooling, which serves to boost engine power and net efficiency, and
- waste heat emitted from the engine surface to the environment as radiation and convection heat.

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While exhaust gas heat is dissipated by gas exchange in the exhaust process, all other waste heat must inevitably be dissipated with the aid of a coolant (water, oil or air).

Heat that accumulates in various points of an engine (Fig. 14-1) is transferred to water as the heat transfer medium for recovery of varying complexity. While cooling energy is transferred to water/water or air/water heat exchangers without any problem, the transfer of exhaust gas heat loaded with particulate matter and soot particulates to a gas/water heat exchanger proves to be somewhat more complicated (see Sect. 9.2.5.5).

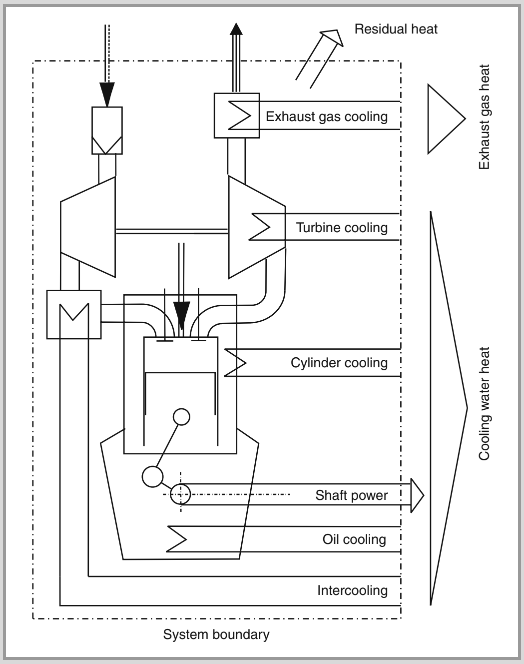

Fig. 14-1 External heat balance and waste heat of a diesel engine

K. Mollenhauer, H. Tschoeke (eds.), *Handbook of Diesel Engines*, DOI 10.1007/978-3-540-89083-6_14,

Radiation and convection heat emitted by an engine is usually dissipated by aerating and ventilating the underhood environment. In principle, it may also be dissipated with the aid of an air/water heat pump and recovered. This has been implemented in very few cases though.

In addition to being dissipated and exchanged differently, the different types of diesel engine waste heat also have different temperature levels corresponding to their place of origin in an engine. The waste heat with the highest temperature level, exhaust gas heat accumulates in the range of 300–500°C depending on the type and size of the engine. Engine cooling water outlet temperatures are usually in the range of 75–95°C. While water temperatures in an intercooler or low temperature intercooler are 30–40°C during multi-stage intercooling, water temperatures in a high temperature intercooler can reach the temperature level of engine cooling water. Temperatures of water used to cool lubricating oil heat are also usually in or slightly below the range of the temperature level of the engine cooling water.

14.1.3 Determination of Waste Heat Outputs

14.1.3.1 Diesel Engine Energy Balance

The following relationships may be referenced to determine a diesel engine's waste heat output.

The following applies to a diesel engine's external heat balance:

$$P_B = P_e + \Phi_A + +\Phi_K + \Phi_R.$$

Accordingly, the heat Φ_{zu} supplied as fuel power P_B corresponds to the product from the fuel mass flow $\dot{m}_B$ and calorific value H_u

$$\Phi_{zu} = P_B = \dot{m}_B \cdot H_u,$$

to the sum of the net (mechanical) power P_e, the heat output Φ_A discharged with the exhaust gas and the cooling capacity Φ_K and to the loss to the environment by radiation and convection contained in the remainder Φ_R.

The overall cooling capacity Φ_K

$$\Phi_K = \Phi_{ZK} + \Phi_{\ddot{O}K} + \Phi_{LLK} \tag{14-1}$$

includes the cooling energy Φ_{ZK} emitted by the engine (cylinders) and the heat fluxes that accumulate in the oil cooler ($\Phi_{\ddot{O}K}$) and intercooler (Φ_{LLK}).

14.1.3.2 Exhaust Heat Output Φ_A

Relative to the system boundary specified by the ambient condition (p_U, T_U), the following, expressed by the enthalpy difference employing the particular specific enthalpy h in (kJ/kg), applies to the exhaust heat output after it exits the turbocharger (index L: air; index A: exhaust gas):

$$\Phi_A = \dot{m}_A h_A - \dot{m}_L h_L = \dot{m}_A[h_A - (1/\delta_0)h_L].$$

With the minimum air requirement L_{min} and the total air/fuel ratio λ_V of combustion, the mass flow ratio $1/\delta_0 = \dot{m}_L/\dot{m}_A$ follows from:

$$1/\delta_0 = L_{min}/(1 + \lambda_v L_{min}). \tag{14-2}$$

Of this, the following is utilizable in the exhaust gas heat exchanger (index AK):

$$\Phi_{AK} = \dot{m}_A \eta_{AWT}(h_{A1} = h_{A2}),$$

where η_{AWT} = 0.95. . .0.98 can be employed as the exhaust gas heat exchanger's efficiency factor and $T_{A1} = T_A - 5$ K or T_{A2} = 160. . .180°C as the exhaust gas temperatures (to prevent wet corrosion). Values for the net enthalpies h of air and exhaust gas relative to the absolute zero point as a function of the temperature and the air/fuel ratio can be gathered from the graph in Fig. 14-2 based on [14-1].

14.1.3.3 Cooling Energy Output Φ_K

The heat an engine dissipates as cooling energy is normally composed of three components (see Eq. (14-1)). Chapter 11 lists guide values for the distribution of the three components in different engines and the valence of the cylinder and lubricating oil cooling energy.

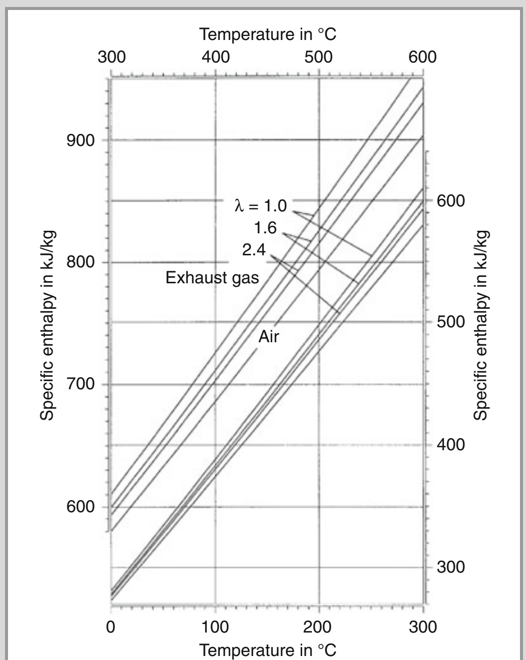

Fig. 14-2 Specific enthalpy of air and exhaust gas as a function of the temperature and the air/fuel ratio

14.1.3.4 Heat Output Φ_{LLK} Dissipated in an Intercooler

Corresponding to the isentropic compression ratio π_L of a "supercharger" compressor, the compression of the air aspirated at ambient temperature T_U increases the temperature of the charge in the compressor outlet to T_{L1} = intercooling inlet temperature. With the compressor's isentropic efficiency η_{SL}, the following ensues for the relative temperature increase T_{L1}/T_U or T_2/T_1 , Eq. (2-37):

$$T_{L1}/T_U = [1 - (\pi_L^{\kappa-1/\kappa} - 1)/\eta_{SL}].$$

With the air temperature T_{L2} in the intercooler outlet, the heat output dissipated in an intercooler is then

$$\Phi_{LLK} = \dot{m}_L(h_{L1} - h_{L2}).$$

With the reference temperature T_U = 298 K according to ISO 3046-1 and the temperature of the charge when it enters the engine $T_{L2} > T_L$, Φ_{LLK} can also be determined with the aid of an h-T diagram (Fig. 14-2).

14.1.3.5 Air and Exhaust Gas Mass Flow Guide Values

With the guide values for the specific air flow rate l_e in kg/kWh from Table 14-1, applying the mass flow ratio δ_0, (Eq. (14-2)) as a function of the increase of the air mass during combustion yields the following:

Table 14-1 Specific air consumption l_e, in kg/kWh

Commercial vehicle diesel engine	With turbocharger and intercooler	l_e = 6.0 .. 6.4
High speed high performance diesel engine	With turbocharger and intercooler	l_e = 6.8 .. 7.2
Medium speed diesel engine	With turbocharger and intercooler	l_e = 7.0 .. 7.2
Low speed two-stroke diesel engine	With turbocharger and intercooler	l_e = 9.8 .. 10.5

$$\dot{m}_A \approx l_e P_e \delta_0.$$

The stoichiometric air/fuel ratio $L_{\min}$ follows from the calculation of combustion in the common elemental analysis of the fuel, applying the following as guide values:

– diesel fuel (DK) $L_{\min}$ = 14.6 kg air/kg fuel or
– heavy fuel oil (HF) $L_{\min}$ = 14 kg air/kg fuel.

Table 14-2 lists components measured at full load relative to the diesel engines' power supplied by the fuel as guide values

Table 14-2 Diesel engine parameters

Parameter		Engine type	
		18V 32/40 MAN Diesel	18V 48/60 MAN Diesel
p_e	bar	24.9	23.2
Bore/stroke	mm/mm	320/400	480/600
Power	kW	9,000	18,900
Speed	rpm	750	500
Exhaust gas temperature	°C	310	315
Percentage of fuel power			
HT cooling water circuit[a]	%	14.2	13.8
LT cooling water circuit[b]	%	10.7	9.8
Exhaust gas (180°C)[c]	%	12.5	12.7
Radiation and convection	%	1.9	1.7
Efficiencies[d]			
η_e[d] (effect.)	%	46.2	47.7
$\eta_{a\ (therm.\ usable)}$[e]	%	37.4	36.3
$\eta_{ges\ (effect.\ +\ thermal)}$[e]	%	83.6	84.0

[a] Includes: cylinder cooling + intercooling
[b] Includes: LT percentage of intercooling + oil cooling
[c] Percentage of exhaust gas heat when cooled to 180°C
[d] Including oil pump(s) without cooling water pumps
[e] Including low temperature heat recovery

for the dissipated and, where applicable, effective heat from diesel engines of different sizes.

14.2 Options of Waste Heat Recovery

14.2.1 Recovering Waste Heat as Mechanical Energy

14.2.1.1 Turbocompounding

While the conversion of waste heat into mechanical energy whenever possible would appear to be an obvious extension of a diesel engine's primary purpose, i.e. to emit mechanical energy, narrow limits are imposed in practice, particularly with regard to cost effectiveness. The impacts on specific engine costs resulting from technical complexity and low conversion efficiency make the use of such processes questionable, particularly in smaller engines. Nonetheless, turbocompounding in which exhaust gas performs additional effective brake work in a downstream exhaust gas turbine, which is either transferred to the output shaft or a generator, is employed in commercial vehicle engines and above all in large engines (see Sects. 2.2.4.4 and 18.4.4).

14.2.1.2 Steam Plant (Bottoming Cycle or Organic Rankine Cycle)

Engine waste heat may also be utilized in steam plants. Also called bottoming cycle plants, they are usually based on the Clausius-Rankine process as an ideal process (Fig. 14-3). According to the Carnot process, the maximum effective temperature interval is subject to narrow limits between the steam temperature achievable with the exhaust gas temperatures and the ambient air as the process waste heat sink. Exhaust gas provides the highest temperatures of 300–500°C (see Tables 14-2 and 1-3). The steam temperatures vary in the range of 200–250°C depending on the particular design (exhaust gas cooling interval). With the exception of partial recovery to preheat feed water with commensurate additional complexity (Fig. 14-3), cooling water heat (engine cooling water, charge air heat, oil heat, etc.) is poorly suited for the generation of mechanical or electrical energy because of its low temperatures.

Continuous advances and increases in effective engine efficiencies and the attendant lower exhaust gas temperatures are increasingly diminishing the option of exhaust gas heat recovery in steam plants.

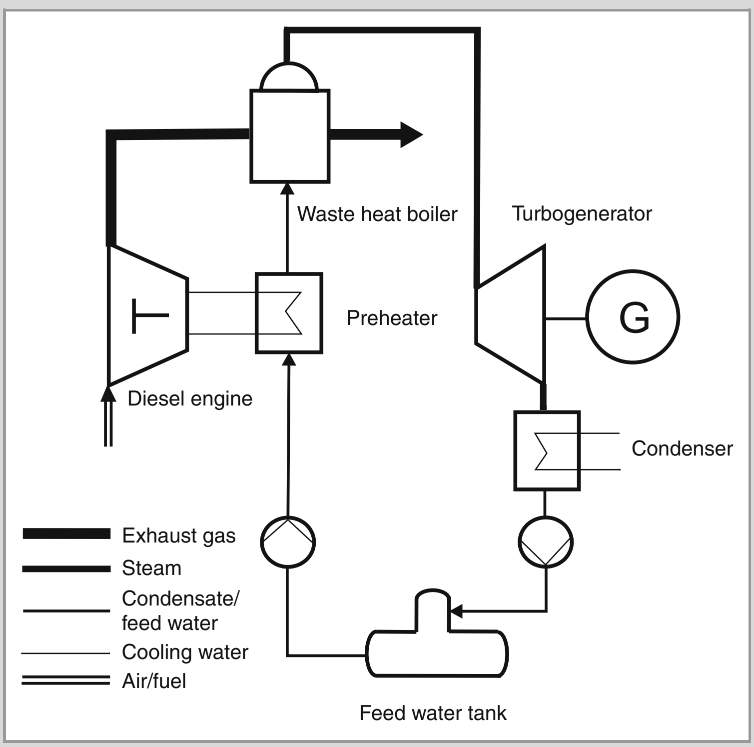

Fig. 14-3
Diesel engine with a downstream steam power process (bottoming cycle) to generate electricity in a turbogenerator

Steam turbines, screw engines and reciprocating steam engines are possible expansion engines. While steam engines with speeds between 750 and 1,500 rpm can directly drive generators, relatively high speed steam turbines and screw engines require gears to adjust speeds to connect to a generator. This additionally diminishes the efficiencies of expansion engines, which are low anyway because of their low outputs.

In addition to steam, fluids with better boiling characteristics than exhaust gas temperatures may also be used as the cycle medium [14-2]. Often called cold vapors or organic vapors, these are common refrigerants that operate in the organic Rankine cycle. Advantages from the better cycle efficiencies anticipated offset disadvantages in terms of toxicity, thermal stability, material compatibility, etc. Steam makes safe operation possible, even with conventional fluorochlorohydrocarbon refrigerants that have a harmful impact on the ozone layer. Detailed tests on a commercial vehicle diesel engine have also corroborated this [14-3]. The maximum increase in output determined was 3%. Although these results demonstrate the limitedness of options for the recovery of waste heat for conversion into mechanical or electrical energy, this process is increasingly attracting interest because of steadily rising fuel prices. Systematic developments would definitely improve the results, above all in large plants [14-4].

In the meantime, the automotive industry is also working on potential applications of this process in passenger cars [14-5].

14.2.2 Recovering Waste Heat as Thermal Energy

14.2.2.1 Heating and Process Heat

Apart from direct utilization in residential heating systems, the technically simplest option for waste heat recovery is the heating of process water. Furthermore, it may be recovered as process heat in manufacturing or to produce fresh water from salt water on ships. Yet, the main application is probably its recovery in motor vehicles to heat the passenger cabin. A vehicle without this is unimaginable today. Classic heating is integrated in the engine coolant circuit. Not only have efficiency-optimized vehicle engines proven to heat passenger compartments poorly in the cold start and warm-up phase because their heat output is inadequate but their pollutant emission, specific fuel consumption and engine wear also surpass that of engines at operating temperature. Hence, so-called auxiliary heaters, fuel-powered heating units that compensate for the heat deficit, are increasingly being implemented in diesel engines in particular [14-6].

14.2.2.2 Cogeneration

General

The simultaneous recovery of mechanical energy and accumulating heat leads to the principle of cogeneration, which aims to recover a maximum of the primary energy supplied by the fuel and thus conserve energy resources as well as take advantage of the attendant reduction of combustion products to reduce the emission of pollutants.

The use of cogeneration plants is both economically and ecologically advantageous.

Combined Heat and Power Stations

According to VDI Guideline 3985 [14-7], combined heat and power stations (CHPS) are cogeneration plants with combustion engines or gas turbines, which generate power and effective heat simultaneously.

A combined heat and power station (Fig. 14-4) consists of one or more CHPS modules with the auxiliary equipment necessary for operation, the related switching and control equipment, noise protection measures, exhaust gas outlets and the appropriate installation space.

The basic unit of a combustion engine CHPS, a CHPS generating set consists of a combustion engine as the generator of mechanical and thermal energy, a generator as the converter of mechanical energy into electrical energy and power transmission and suspension elements. Together with the heat exchange components, the control and monitoring systems, the intake and exhaust system, the lubricating oil and fuel system and the safety systems, it forms a CHPS module.

While, CHPS generating sets for large outputs are usually delivered to construction sites and all other components are custom finished for the systems, compact modules (Fig. 14-5) containing every component including the primary exhaust silencer are usually employed for smaller outputs.

Combined heat and power stations are implemented in municipal facilities such as hospitals, swimming pools and schools and in industries and businesses as well as in office and residential buildings. Their electrical output extends from the low kilowatt to two digit megawatt range. While gas turbines are frequently implemented as drive engines when outputs are large and CHPS heat is recovered to generate steam, internal combustion engines are used when units and outputs are smaller. In addition to diesel engines and dual fuel engines, spark ignited engines are predominantly employed because their exhaust gas emits few pollutants (see Sect. 4.4).

In the context of the use of renewable energy sources, vegetable oils, particularly rape oil or rape oil methyl ester (RME), are available as fuel for diesel engines and dual fuel engines [14-8].

A number of laws passed in the German Bundestag have further improved the cost effectiveness of CHP plants in Germany.

The law introducing the Ecological Tax Reform exempts CHPS with outputs of up to 2 MW from the electricity tax and the mineral oil tax when their annual utilization ratio is at least 70%. The Act on Promoting the Generation of Electricity from Renewable Energy Sources establishes lucrative feed-in

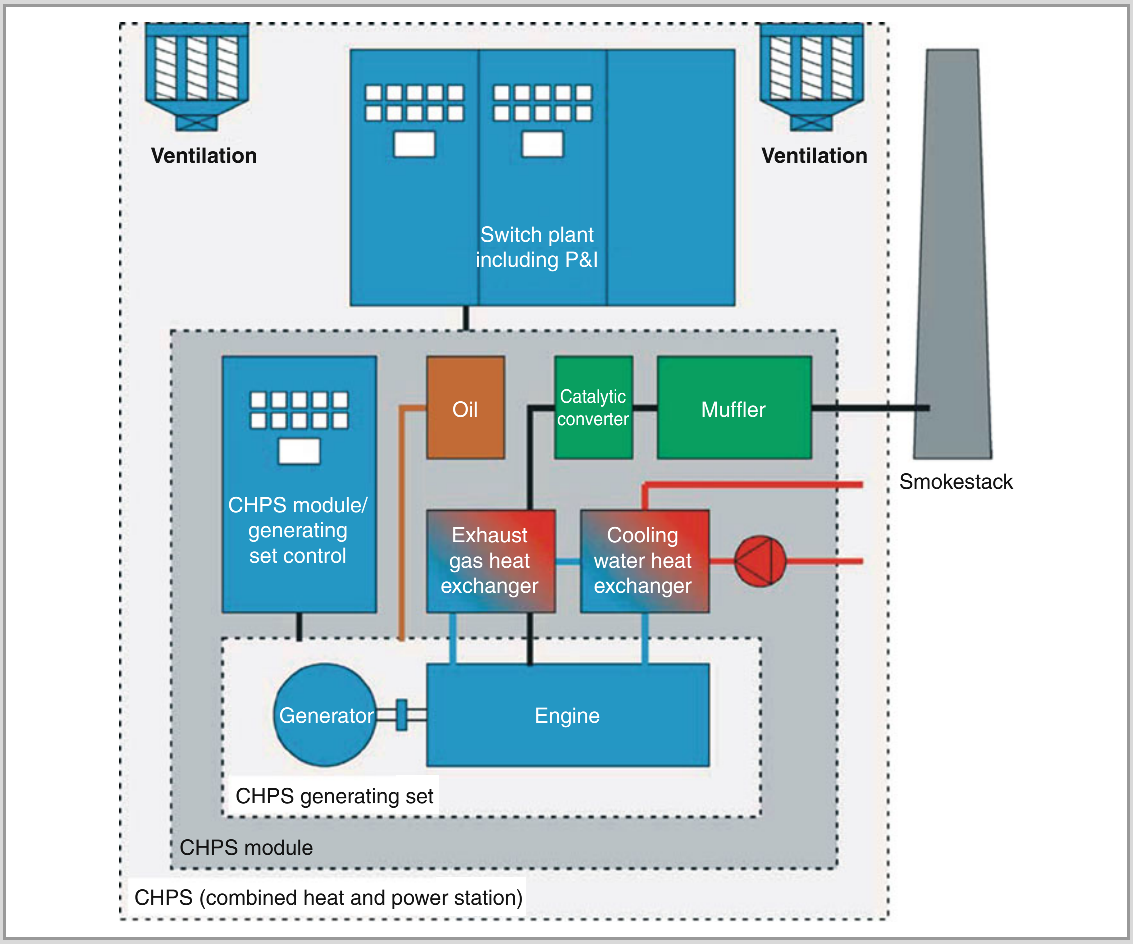

Fig. 14-4 Definition and specification of CHPS components according to DIN 6280

tariffs. A bonus in addition to the feed-in tariff is also awarded when fuels made from renewable raw materials are used.

Combined heat and power generation in CHPS indisputably saves a more substantial share of primary energy than the separate generation of power in a power plant and heat in a boiler. In terms of NO_X and CO_2, the input of pollutants into the atmosphere is lower than the separate generation of power in a thermal power plant and heat in a boiler.

However, the prerequisite for this is the implementation of appropriate measures that reduce NO_X emissions. The limits of the Technical Instructions on Air Quality Control (*TA Luft*) have to be observed when combined heat and power stations with thermal firing capacities above 1 MW are operated in Germany (see Sect. 15.2). While in-engine measures (lean burn systems) help spark ignited engines considerably undershoot the limits that are already lower than for diesel engines, exhaust emission control systems are essential for the operation of diesel and dual fuel engines in the majority of cases.

An economically and ecologically trendsetting concept, a combined heat and power station consisting of four CHPS modules powered with the renewable fuel rape oil methyl ester supplies the Reichstag building in Berlin with power.

Consequently, the potential savings in CO_2 is taken advantage of in two ways, namely by burning a renewable fuel known to have a good CO_2 balance as well as by applying the principle of cogeneration with its inherent potential to save primary energy and thus emit little CO_2.

With 400 kW of electrical power output apiece at electrical efficiencies of 42.5%, the four CHPS modules recover 90% of the primary energy. Heat is decoupled in an HT circuit (engine cooling water – heat + oil cooler heat + exhaust gas heat) at 110°C and an LT circuit (charge air cooling energy) at 40°C.

The units were provided with a system for exhaust gas aftertreatment to – as the client demanded – minimize not only CO_2, which is admittedly nontoxic but damaging to the earth's atmosphere, but also exhaust gas pollutants formed

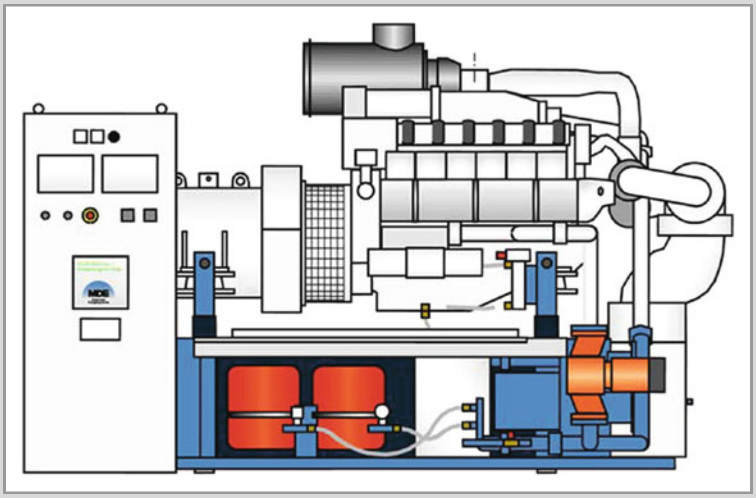

Fig. 14-5
Compact design of a CHPS module

during combustion. The nonvolatile particles are separated out of the exhaust gas in a particulate filter by means of catalytically coated filter cartridges. An SCR catalyst consisting of coated honeycombs reduces the NO_X emission by spraying in urea and an additional oxidation catalyst reduces the emission of carbon monoxide and hydrocarbons.

Thus, the mandated emission values, which are far lower than in *TA Luft* (see Sect. 15.2), are observed:

– soot (particulates) < 10 mg/m^3
– particulate matter < 20 mg/m^3
– nitrogen oxides $NO_X < 100$ mg/m^3
– carbon monoxide CO < 300 mg/m^3
– hydrocarbons HC < 150 mg/m^3

(According to *TA Luft*, based on the standard level (273.15 K; 101.3 kPa) after deducting the moisture content in exhaust gas with an oxygen content of 5%, 4,000 mg/m^3 of NO_X were permissible at the time of construction!). Even though experiences are available from a large number of successfully operating CHP plants, the expedience of CHPS implementation and the most promising concept have to be evaluated in every individual case.

Resource conservation and environmental relief alone rarely suffice to sway an operator to make a positive buying decision. Therefore, like other capital goods, the cost effectiveness of CHP must also be subjected to analysis. However, a plant must be designed before a feasibility study is performed. The current situation of the energy flows, the energy supply and procurement contracts must be reviewed. This requires detailed data on the power and heat demand over time. The operating hours of the CHPS modules can be determined by entering module power based on annual load duration curves and daily and weekly load curves [14-9]. Naturally, the mode of CHPS operation is also entered. Is it operated in heat driven, power driven or alternating mode? Is running it at peak load advantageous based on the power procurement contract? What does the energy demand forecast look like?

Once the concept has been established, a feasibility study can be performed, applying methods of investment mathematics familiar from capital expenditure budgeting. Along with the annuity method, the net present value method has particularly proven itself.

The curve of present value over years of useful life not only delivers the amortization period but also the return at the end of the useful life. Along with information on cost effectiveness, the amortization period, which specifies the time span between the time of investment and the time at which the higher capital investment is recouped through savings of energy costs, represents an important parameter for the assessment of the financial risk. The shorter the amortization period, the lower the investment risk is. Both information on the risk and data on the achievable surplus provide support for a decision on the appropriate concept.

Parameters of CHP Plants

Part 14 of the standard DIN 6280 defined important parameters in order to establish uniform rules for the many terms employed over the years, (see Table 14-3) [14-10]. Issued in August 1997, the standard applies to combined heat and power stations (CHPS) with reciprocating internal combustion engines, which generate alternating current and useful heat.

Part 14 of DIN 6280 defines utilization ratios similarly to efficiencies. The generated energy (electrical, thermal and

Table 14-3 Definition of CHPS efficiency according to DIN 6280

Electrical efficiency	η_{el}	Ratio of the true electrical power output generated to the heat input from the fuel supplied based on the calorific value (Hu)
Thermal efficiency	η_{th}	Ratio of the generated thermal power to the heat input from the fuel supplied based on the calorific value (Hu)
Overall efficiency	η_{ges}	Sum total of the electrical efficiency and thermal efficiency. The overall efficiency does not account for the power for the auxiliary drives.

total) is set in relation to the thermal energy of the quantity of fuel supplied relative to the calorific value (H_u) within a longer period (e.g. one year). Unlike efficiencies, utilization ratios also incorporate the energy of auxiliary drives (e.g. pumps and fans) and downtime losses.

Each of the measured or specified efficiencies (see Table 14-4) depend on the operating state of the CHPS (rated load or part load, speed, cooling water temperature, charge air temperature, cooling temperature of the exhaust gas, etc.).

All parameters, efficiencies and utilization ratios for a particular unit (generating set, module or CHPS) relate to a defined system boundary, e.g. current output at the generator terminals, hot water inlets and outlets in the module and charge air cooling water inlets and outlets in the module. These boundary conditions are indispensible for the classification of the parameter data.

While the measurements or specifications of efficiencies relate to constant operating conditions, startup and shutdown processes, part load operation and downtimes are also incorporated in the data of utilization ratios. Consequently, the planning and design of a complete plant and the mode of operation selected by the operator substantially influence a plant's utilization ratios. Hence, it is impossible to draw any conclusions about a plant's quality from the utilization ratios.

The desire to prolong the operating hours of emergency diesel units by attaching process equipment to supply heat from cogeneration or to even run them in continuous operation has been expressed repeatedly as combined heat and power stations with gas engines have spread. This is absolutely inadvisable since diesel emergency power plants are solely designed for emergency use. This pertains to engines engineered for such operation (higher output at low operating hours) and every other component not intended for continuous operation. An emergency power plant can only be operated in parallel operation with the grid or as an emergency power supply system after relatively extensive modifications (engine, control, etc.) or a reconfiguration of the electrical output and the attachment of every component necessary to supply heat from cogeneration as well as the exhaust gas aftertreatment system needed to comply with the legally mandated limits for pollutant emissions.

Table 14-4 Ranges of common CHPS efficiencies

Electrical efficiency	η_{el}	25–48%
Thermal efficiency	η_{th}	30–56%
Overall efficiency	η_{ges}	65–92%

Diesel Engine Heat Pump

A diesel engine heat pump is a compression heat pump with a compressor driven by a diesel engine. Figure 14-6 is a schematic of the principle of diesel engine heat pump design. Piston compressors, screw compressors and turbocompressors are the types of compressors predominantly implemented to compress the working medium (refrigerant). The compressed and simultaneously heated working medium flows into a condenser where it emits useful or heating heat at a high temperature level. After the pressure in the throttle drops, the working medium in the evaporator absorbs energy in the form of unusable heat from the ambient air, river water, salt water or other low temperature heat sources.

In accordance with the principle of a heat pump, the addition of compressor work converts low temperature heat into high temperature heat.

A ratio of the heat output $\dot{Q}_c$ emitted in the condenser to the compressor input power P_v where $P_v \equiv P_e$ applies, the heat coefficient of performance

$$\varepsilon = \dot{Q}_c / P_e$$

serves as a parameter for the evaluation of the heat pump process.

Dependent on the temperature of the heating heat and the low temperature heat from the refrigerant fed to the evaporator, the coefficients of performance for heat pumps commonly implemented for residential or process water heating have values between 2 and 4.

However, a diesel engine heat pump also recovers the thermal energy emitted by the engine. It may either be supplied to the heating circuit after the condenser to increase the

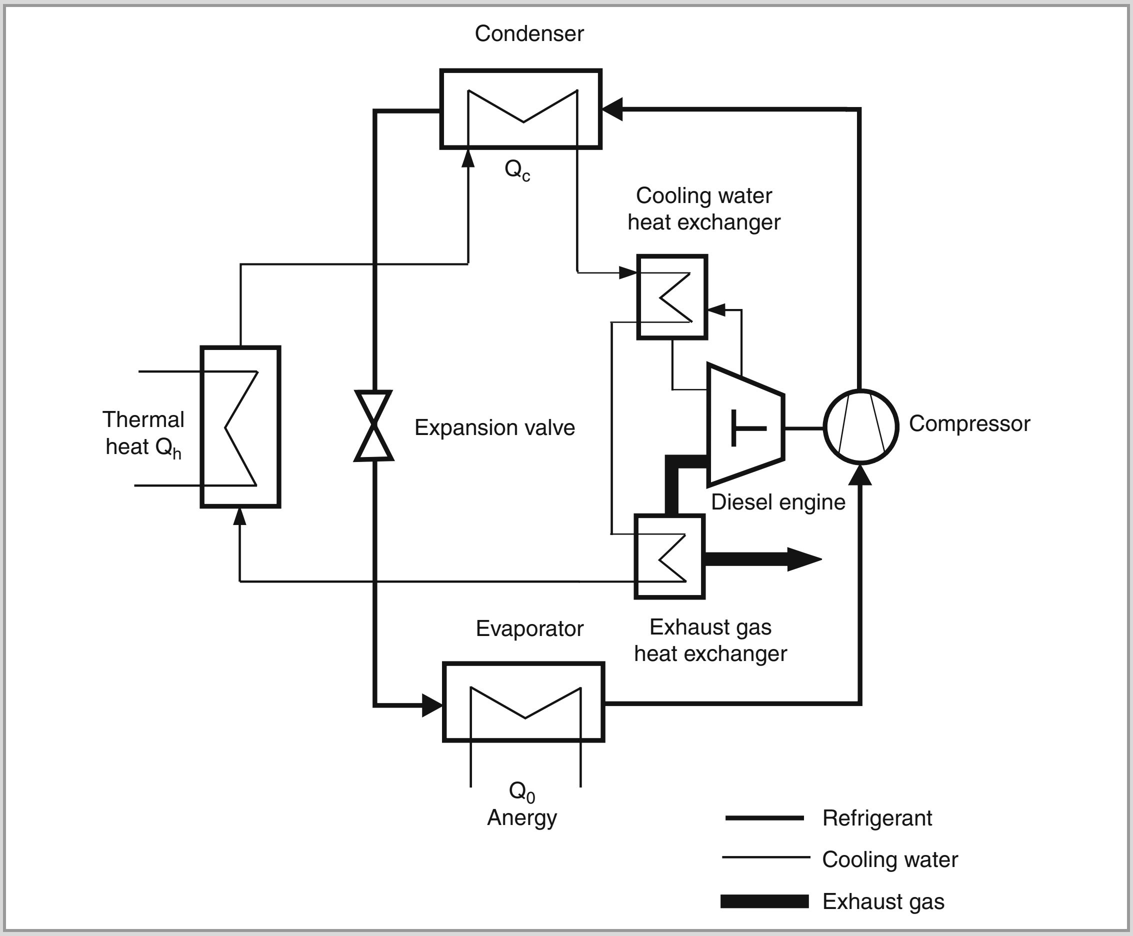

Fig. 14-6 Schematic of a combustion engine heat pump

flow temperature or drawn into a second separate circuit to supply other consumers.

A ratio of the total useful heat output $\dot{Q}_N$ to the energy input of the fuel P_B based on the calorific value (H_u), the heat factor

$$\zeta = \dot{Q}_N / P_B$$

serves as a parameter for the evaluation of the overall process of a diesel engine heat pump.

The heat factor can also be calculated from the relationship

$$\zeta = \varepsilon\eta_e + \eta_a,$$

i.e., from the coefficient of performance, the engine efficiency and the percentage of useful engine waste heat relative to the fuel power supplied.

Dependent on the temperature of the heating heat and the low temperature heat from the refrigerant supplied to the evaporator and the corresponding engine data (see also Table 14-2), the heat factors for heat pumps commonly implemented for residential or process water heating have values between 1.5 and 2.

Figure 14-7 presents the energy balance of a diesel engine heat pump system designed with an engine power of 250 kW and a heat output of 1,085 kW.

Heat pumps utilized for residential and process water heating emit heating heat with temperatures of 70°C/50°C (forward flow/return). Groundwater with a temperature of 10°C is used as the energy source. When the water is cooled down by 4 K, evaporator temperatures of +1°C can be run at full load and 4–5°C at part load operation. Crucial for the

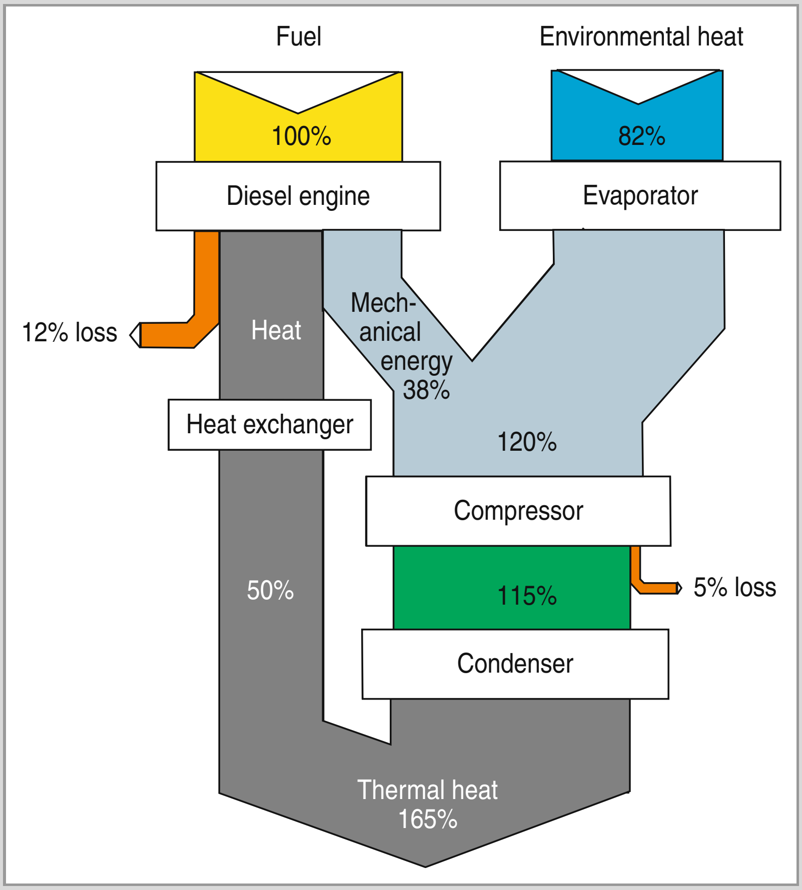

Fig. 14-7
Energy flow diagram of a diesel engine heat pump

heat pump's coefficient of performance, a temperature lift of 50–59 K takes effect between refrigerant evaporation and condensation at condenser temperatures of 60°C at full load; values are correspondingly lower at part load operation.

Compressor Module

The operation of compressors driven with internal combustion engines furnishes another option for direct waste heat recovery. The units employed to generate compressed air are predominantly screw compressors and turbocompressors.

The design of a compressor module is comparable to a combined heat and power station module with a compressor instead of a generator.

As the energy flow diagram in Fig. 14-8 indicates, 87% of the primary energy may be utilized with a diesel engine compressor module in the 400 kW performance class.

14.2.2.3 Trigeneration

In the majority of cases of application, CHP plants are designed for heat driven operation, i.e. the output is specified so that CHPS heat can be recovered the entire year if possible. This results in high operating hours and guarantees cost effective operation. In cases of application with heat recovery for residential heating systems, this leads to CHPS output designed more for the lower heat demand arising in summer. Hence, planning should entail verifying that cooling is not required in the summer months instead of heat. Administrative buildings with larger EDP facilities, hospitals, hotels, shopping centers, etc. especially require cooling and air conditioning in many cases. A plant based on the principle of combined heating, cooling and power (CHCP) can definitely be operated cost effectively when the annual demand for electrical energy, thermal heat and cooling energy resembles the curve in Fig. 14-9.

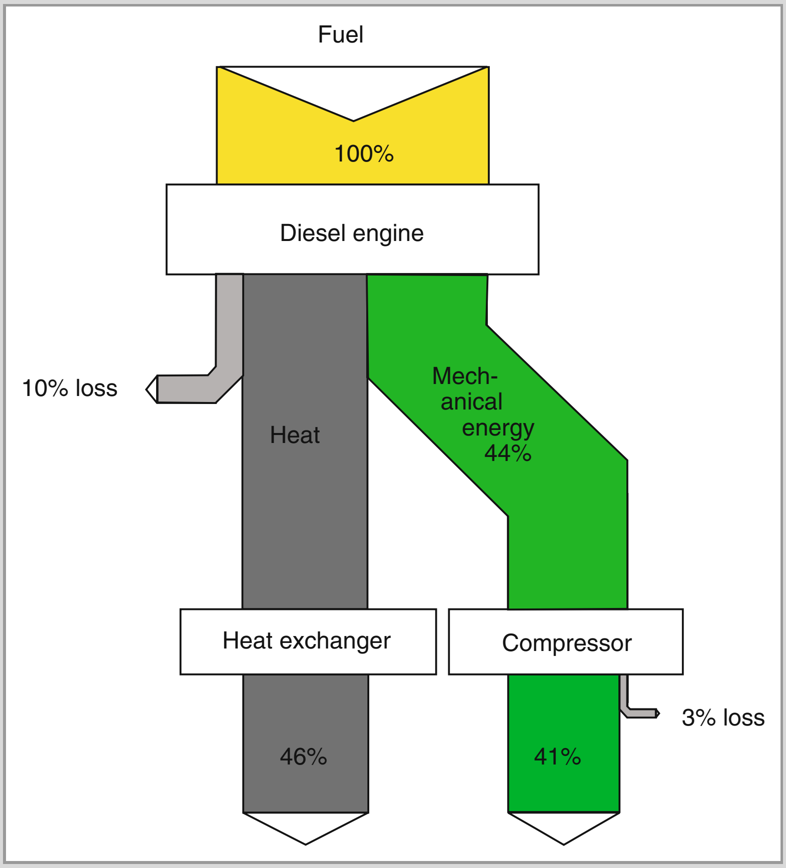

Fig. 14-8
Energy flow diagram of a compressor module

A plant based on the principle of trigeneration consists of one or more combined heat and power station modules coupled with a sorption refrigerator. For the most part, absorption refrigeration units are utilized. Adsorption refrigeration units are also utilized in some cases [14-11].

Mainly water is used as the refrigerant and lithium bromide as the solvent in air conditioning systems (cold water circuit 12°C/6°C). Mainly ammonia is used as the refrigerant and water as the solvent in cooling systems below 0°C.

Figure 14-10 presents the energy balance of a CHCP plant with an absorption refrigeration unit at CHPS heating water temperatures of 95°C/85°C, a cooling tower temperature of 27°C for absorber cooling and cold water temperatures of 6°C/12°C.

The advantage of any sorption unit is its low wear (only a few moving parts) and thus the low maintenance required, its good part load characteristics with infinitely variable load control and its low noise emission. Since they do not require fluorochlorohydrocarbon refrigerants that contribute to the greenhouse effect and adversely impact the ozone layer sorption units, also have advantages over compression refrigeration units in terms of environmental compatibility.

Examining the expedience of a cogeneration or trigeneration plant and identifying the most promising concept is fundamentally necessary in every individual case.

Ecological benefits constitute a sufficient basis for a decision to build a plant in only the rarest of cases. The investment required makes meticulous plant planning and a detailed evaluation of cost effectiveness essential.

14.2.3 Concluding Remarks

Although established types of mechanical and thermal energy recovery from diesel engine waste heat have often failed to satisfy the criteria of cost effectiveness in the past, these technologies are increasingly attracting attention again as fuel prices continue to rise. Moreover, the high level of fuel prices expected in the future too will also advance the use of renewable energy sources in combustion engines. This will be financially advantageous, not least because of lower taxes. Particularly from the

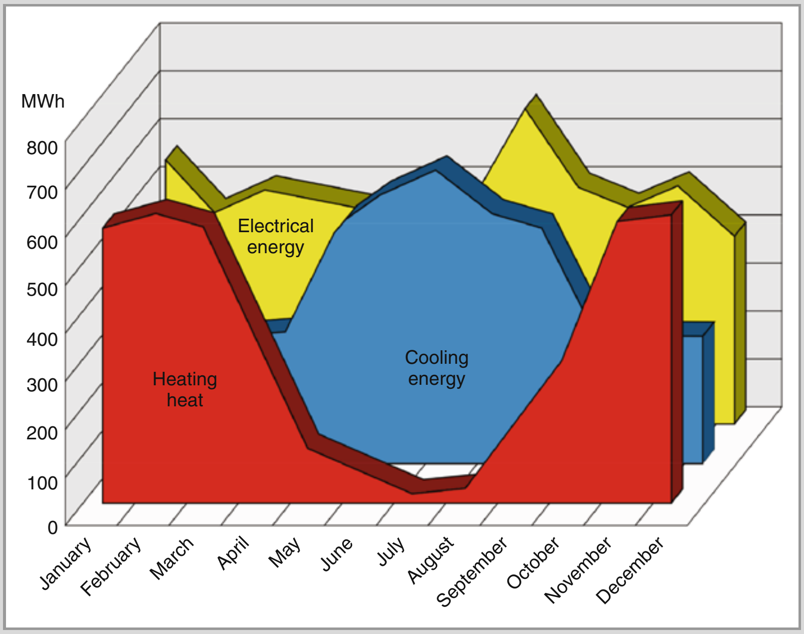

Fig. 14-9
Energy profile of electrical energy, heating heat and cooling energy

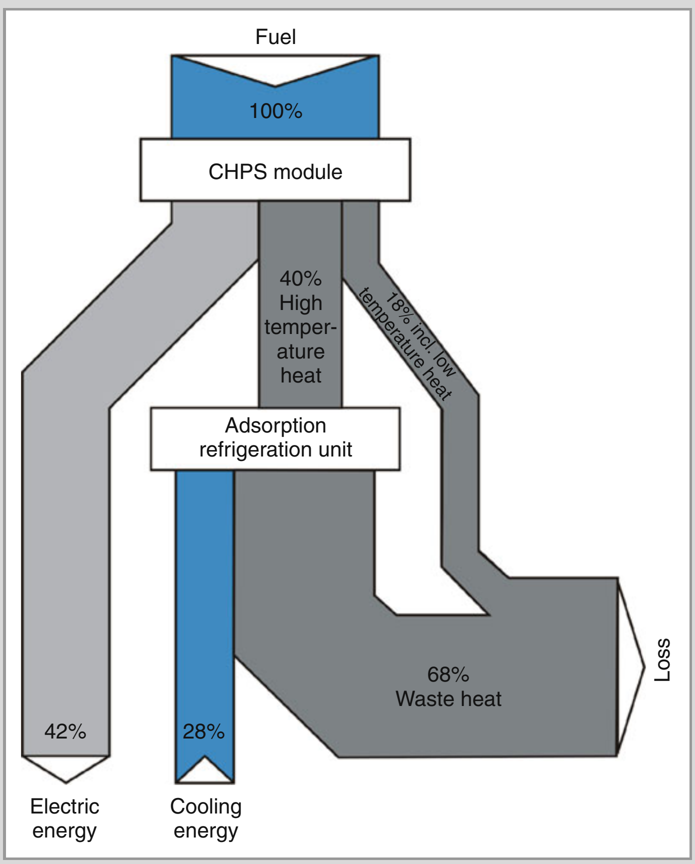

Fig. 14-10
Energy flow diagram of a trigeneration plant

perspective of conserving fossil energy sources and protecting the environment by lowering the input of CO_2 into the atmosphere, the recovery of diesel engine waste heat, especially in combination with the use of biogenic energy sources, will continue to grow in importance in the future.

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